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[54] EXHAUST PRESSURIZATION OF LOAD RESPONSIVE SYSTEM

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Related U.S. Application Data

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[51] Int. Cl.[3] F15B 13/08

[52] U.S. Cl. 137/596.13; 60/428; 60/452; 60/468; 60/486; 91/436

[58] Field of Search 91/436; 137/596.13; 60/428, 452, 468, 486

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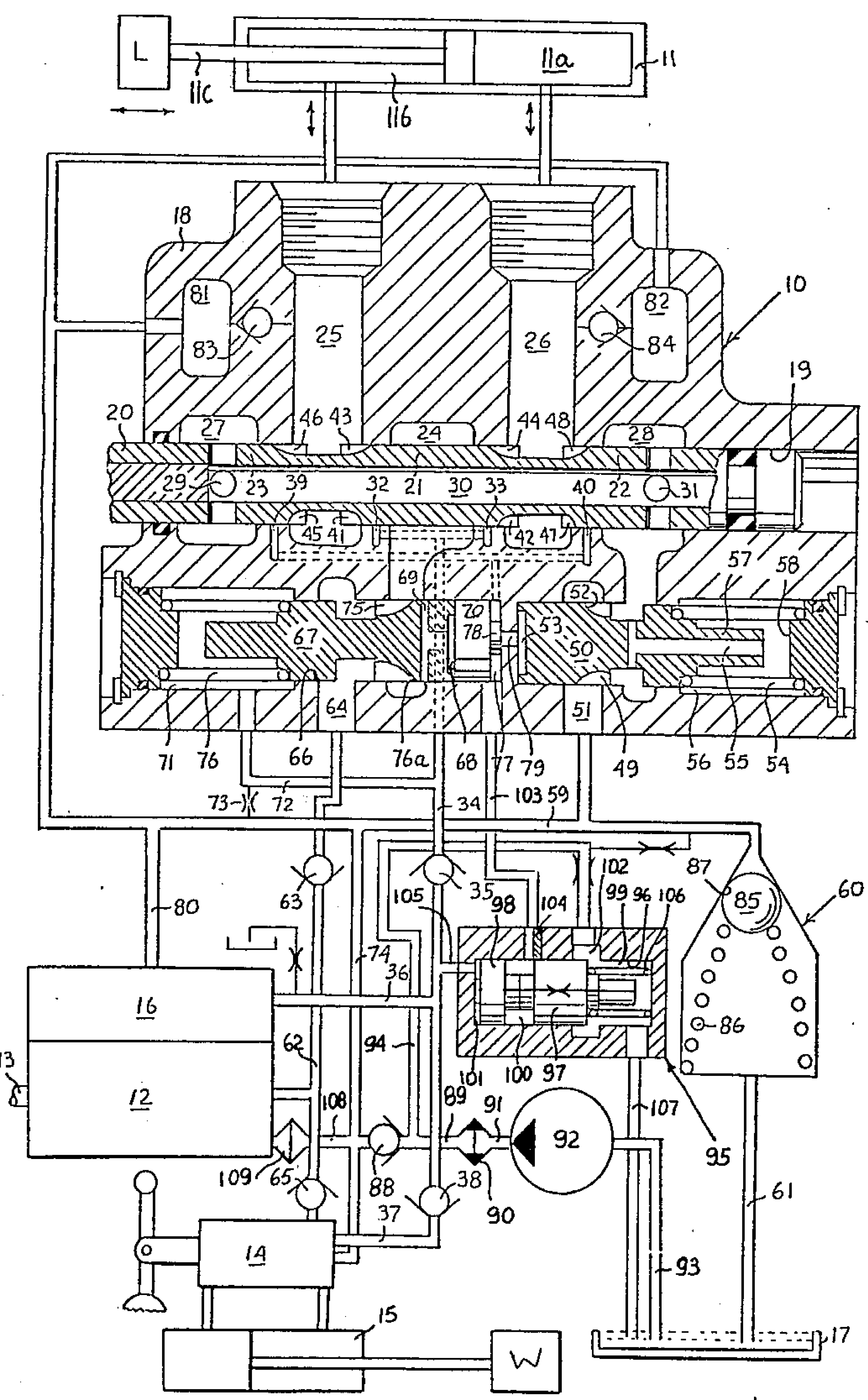
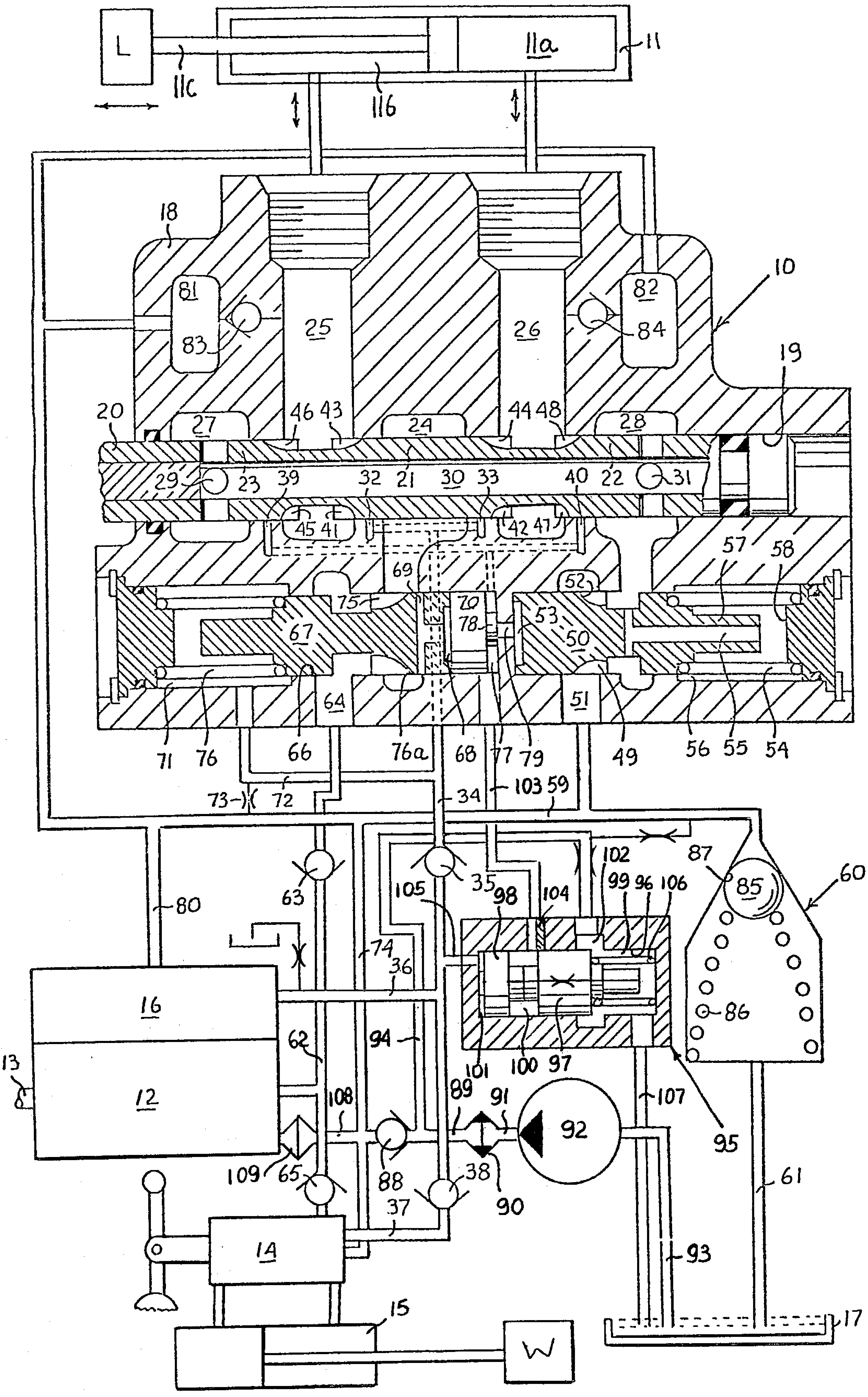
[57] ABSTRACT

In a load responsive fluid power and control system employing plurality of control valves with positive and negative load compensation, a closed loop pressurized exhaust system supplied with a make-up and fluid exchange pump equipped with an unloading device responsive to positive and negative load system pressures.

10 Claims, 1 Drawing Figure

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EXHAUST PRESSURIZATION OF LOAD RESPONSIVE SYSTEM

This application is a continuation in part of application Ser. No. 049,660, filed June 18, 1979, for "Exhaust Pressurization Of Load Responsive System," now U.S. Pat. No. 4,249,570, and Ser. No. 960,767, filed Nov. 15, 1978, for "Load Responsive Control Valve", now U.S. Pat. No. 4,209,039.

BACKGROUND OF THE INVENTION

This invention generally relates to a fluid power and control system, in which the exhaust flow of system motors is used directly to provide inlet flow requirement of the system pump.

In still more particular aspects this invention relates to pressurized exhaust system of a plurality of fluid motors, controlled by load responsive valves, the exhaust system supplied with a make-up and fluid exchange pump and used to supply the inlet flow requirements of the system pump.

In still more particular aspects this invention relates to a load responsive fluid power and control system in which the exhaust flow of system motors, supplemented by fluid flow from a make-up and fluid exchange pump, equipped with load responsive unloading and bypass device, is used to supply pressurized inlet flow requirement of the system pump.

Pressurization of the exhaust flow from compensated load responsive valves and also pressurization of system pump inlet is very desirable, but it suffers from the disadvantage of comparatively large throttling losses, directly affecting system efficiency and from the necessity of providing full flow inlet pressurizing pump which is expensive. Closed loop systems, involving a pump and a rotary type fluid motor are well known in the art. Such closed loop systems are characterized by their pressure flow being approximately equal to their exhaust flow. A small exhaust feeding or make-up pump is usually provided to pressurize the exhaust loop and to provide a measure of fluid exchange, in order to cool the closed loop system. Such a make-up pump supplies full flow of pressurized fluid in system standby condition and is not intended for systems, in which comparatively large differences between pump outlet flow and system return flow can take place, which is the case in systems using cylinders as fluid motors. Full flow inlet pressurization has been used in the past, but as previously mentioned it is expensive, since it requires not only a full flow low pressure make-up pump, but also separate drive, large fluid lines, fittings etc.

SUMMARY OF THE INVENTION

It is therefore a principal object of this invention to reduce total pump output by blocking off its flow during control of negative loads, reduce exhaust fluid flow fluctuations, use the system exhaust flow to supply the inlet flow requirement of the system pump and to supplement, pressurize and partially exchange for system cooling purposes, the exhaust system flow by a separate low pressure make-up and fluid exchange pump.

It is a further object of this invention to close the loop of a central load responsive system using plurality of rotary and reciprocating fluid motors, to provide both the necessary pressurization of the exhaust of the compensated load responsive valves and to provide a pressurized inlet flow of the system pump, with minimum

power loss and with use of small make-up pump for exhaust pressurization and flow exchange.

It is a further object of this invention to unload the make-up pump with the load responsive system in its standby condition, and during control of negative load from the piston end of the cylinder, to further minimize the system loss.

Briefly the foregoing and other objects of this invention are accomplished by pressurizing the exhaust circuit of the load responsive system valves without throttling process, while simultaneously providing the advantage of inlet pressurization of system pump. The comparatively small make-up pump, provided for pressurization and cooling of the exhaust loop, is automatically unloaded with the system in its standby condition. The load responsive pressure compensated system valves do not use pump flow during control of negative loads, further reducing the size of the make-up pump and the amount of cooling exchange flow necessary to cool the system.

Additional objects of this invention will become apparent when referring to the preferred embodiment of the invention as shown in the accompanying drawing and described in the following detailed description.

DESCRIPTION OF THE DRAWING

The drawing shows a sectional view of an embodiment of a flow control valve having positive and negative load compensation and exhaust system unloading valve with lines, system flow control, system pump, second load responsive valve, exhaust relief valve, inlet charging pump and system reservoir shown diagramatically.

DESCRIPTION OF THE PREFERRED EMBODIMENT

Referring now to the drawing, an embodiment of a flow control valve, generally designated as **10**, is shown interposed between diagramatically shown fluid motor having chambers **11***a* and **11***b* and piston rod **11***c*, connected to a load L and a pump **12** of a fixed displacement or variable displacement type driven through a shaft **13** by a prime mover not shown.

Similarly, a flow control valve **14**, identical to flow control valve **10**, is interposed between a diagramatically shown fluid motor **15** driving a load W and the pump **12**. Fluid flow from the pump **12** to flow control valves **10** and **14** is regulated by a pump flow control **16**. If pump **12** is of a fixed displacement type, pump flow control **16** is a differential pressure relief valve, which, in a well known manner, by bypassing fluid from the pump **12** into the exhaust circuit, maintains discharge pressure of pump **12** at a level, higher by a constant pressure differential, than load pressure developed in fluid motor **11** or **15**. If pump **12** is of a variable displacement type pump flow control **16** is a differential pressure compensator, well known in the art, which by changing displacement of pump **12** maintains discharge pressure of pump **12** at a level, higher by a constant pressure differential, than load pressure developed in fluid motor **11** or **15**.

The flow control valve **10** is a fourway type and has a housing **18** provided with a bore **19** axially guiding a valve spool **20**. The valve spool **20** is equipped with lands **21**, **22**, **23** which in neutral position of the valve spool **20**, as shown in the drawing, isolate a fluid supply chamber **24**, load chambers **25** and **26** and outlet chambers **27** and **28**. The outlet chamber **27** is connected through ports 29, central passage 30 in valve spool 20 and ports 31 to the outlet chamber 28.

Positive load sensing ports 32 and 33, located between load chambers 25 and 26 and the supply chamber 24 and blocked in neutral position of valve spool 20 by land 21, are connected through signal passage 34, a check valve 35 and signal line 36 to pump flow control 16. In a similar manner positive load sensing ports of flow control valve 14 are connected through line 37, a check valve 38 and signal line 36 to the pump flow control 16. Negative load sensing port 39 is located between load chamber 25 and outlet chamber 27. Similarly, negative load sensing port 40 is located between load chamber 26 and outlet chamber 28.

The land 21 of the valve spool 20 is equipped with signal slots 41 and 42, located in plane of positive load sensing ports 32 and 33 and metering slots 43 and 44, which, in a well known manner, can be circumferentially spaced in respect to each other and in respect to the signal slots 41 and 42. The land 23 is equipped with signal slot 45, located in plane of negative load sensing port 39 and circumferentially spaced metering slot 46. The land 22 is equipped with signal slot 47, located in plane of negative load sensing port 40 and circumferentially spaced metering slot 48. Signal slots 41, 42, 45 and 47, in a well known manner, can be substituted by end surfaces of lands 21, 22 and 23. A suitable device is provided to prevent relative rotation of the spool 20 in respect to bore 19.

The outlet chamber 28 is connected through slots 49, of a negative load control spool 50, to an exhaust chamber 51. The negative load control spool 50 having slots 49, provided with throttling edges 52, projects into control space 53 and is biased towards position, as shown, by spring 54. The negative load control spool 50 is provided with passage 55 connecting the outlet chamber 28 with space 56 and is equipped with stop 57, limiting its displacement against surface 58. The exhaust chamber 51 in turn is connected to exhaust line 59. An exhaust relief valve, generally designated as 60, communicates exhaust line 59 through line 60 to the reservoir 17.

The pump 12 through its discharge line 62 and load check 63, is connected to a fluid inlet chamber 64. Similarly, discharge line 62 is connected through load check valve 65 with the inlet chamber of the fluid control valve 14. The control bore 66 connects the fluid inlet chamber 64 with the fluid supply chamber 24. The control spool 67, axially slidable in control bore 66, projects on one end into space 68, connected to the fluid supply chamber 24 by passage 69 and abuts against a free floating piston 70. The control spool 67 on the other end projects into control space 71, which is connected by passage 72 with positive load sensing ports 32 and 33 and through leakage orifice 73 to exhaust line 59 and to upstream of exhaust relief valve 60. Similarly, control space and leakage orifice of the control valve 14 is connected by line 74 to upstream pressure of exhaust relief valve 60. The control spool 67 is provided with slots 75 terminating in throttling edges 76a, positioned between the inlet chamber 64 and the supply chamber 24. The control spool 67 is biased by a control spring 76 towards position, in which slots 75 connect the fluid supply chamber 24 with the fluid inlet chamber 64.

The free floating piston 70 on one end is subjected to pressure in space 68, which is connected to the fluid supply chamber 24 and on the other end is subjected to pressure in control space 77, which is connected to negative load pressure sensing ports 39 and 40. Projection 78 of the free floating piston 70, in the position as shown, effectively seals port 79 and control space 53 from control space 77.

The exhaust relief valve, generally designated as 60, is interposed between combined exhaust circuits of flow control valves 10 and 14, including bypass circuit of pump 12 and reservoir 17. The pressurized exhaust circuit of flow control valve 10 includes exhaust line 59 connected to bypass line 80 and connected to chambers 81 and 82, which are operationally connected for one way fluid flow by check valves 83 and 84 with load chambers 25 and 26. The exhaust relief valve 60 is provided with a throttling member 85, biased by a spring 86 towards engagement with seat 87.

A check valve 88 is located between line 74, connected to the combined exhaust circuits of flow control valves 10 and 14 and an exhaust make-up circuit, composed of line 89, a heat exchanger 90, line 91 and make-up pump 92, which is connected by line 93 with the reservoir 17. Line 89 is connected by line 94 to an exhaust unloading valve, generally designated as 95. The exhaust unloading valve 95 is provided with bore 96, slidably guiding a spool 97 and a piston 98, which functionally divide bore 96 into spaces 99, 100 and 101. Bore 96 is provided with annular space 102, connected through line 94 with the make-up pump 92. Space 100 is connected by line 103 with control space 77, which communicates directly with negative load sensing ports 39 and 40 or is plugged by a plug 104. Space 101, through line 105, is connected to signal line 36. The spool 97 and the piston 98 are biased by a spring 106 towards position, as shown in the drawing, maintaining communication between annular space 102 and space 99, which is connected by line 107 with system reservoir 17. The make-up pump 92 is connected through line 91, the heat exchanger 90, line 89, the check valve 88, line 108 and schematically shown filter 109 with inlet of the pump 12. Line 89, between the check valve 88 and the filter 109, is connected to line 74 and therefore to the exhaust circuits of the flow control valves 10 and 14.

If the pump 12 is of a fixed displacement type excess pump flow from the differential pressure relief valve or pump flow control 16 is delivered through line 80 to the exhaust line 59 and therefore to the total pressurized exhaust circuit of flow control valves 10 and 14.

The sequencing of the lands and slots of valve spool 20 preferably is such that when displaced in either direction from its neutral position, as shown in FIG. 1, one of the load chambers 25 or 26 is first connected by signal slots 41 or 42 to the positive load sensing port 32 or 33, while the other load chamber is connected by signal slots 45 or 47 to the negative load sensing port 39 or 40, while the load chambers 25 and 26 are still isolated from the supply chamber 24 and the outlet chambers 27 and 28. Further displacement of the valve spool 20 from its neutral position connects load chamber 25 or 26 to the supply chamber 24 through metering slots 43 or 44, while connecting the other load chamber through metering slots 46 or 48 with one of the outlet chambers 27 or 28.

Referring now to the drawing, with the pump 12 of a fixed displacement type started up, the pump flow control 16 will bypass all of the pump flow through line 80 to exhaust line 59. From exhaust line 59 the flow will be supplied to inlet of the pump 12 through line 108 and the filter 109. This flow will be supplemented by the flow delivered from the make-up pump **92** through the heat exchanger **90**, line **89** and check valve **88**. The pump **12** will automatically maintain pressure in discharge line **62** at a constant pressure, higher by a constant pressure differential, than pressure in signal line **36** or pressure in exhaust line **59**. The pressure in exhaust line **59**, due to internal leakage of the pump **12**, will be dictated by the discharge pressure of the make-up pump **92**, discharge port of which is connected through line **94**, annular space **102**, space **99** and line **107** to the system reservoir **17** and therefore is maintained at a low pressure level. When using a fixed displacement pump **12**, for circuit cooling purposes, exhaust line **59**, in a well known manner, may be connected by a suitable leakage orifice with line **94**, as shown in the drawing. Therefore all of the pump flow is diverted by the pump flow control **16** to the low pressure exhaust circuit, as previously described, without being used by flow control valves **10** and **14**. Since signal line **36** is connected by passage **72** with control space **71**, which is also connected through leakage orifice **73** to exhaust line **59**, the bypass pressure in the discharge line **62** will be higher, by a constant pressure differential, than the pressure in exhaust line **59**. This pump bypass pressure transmitted through passage **69** to space **68** reacts on the cross-sectional area of control spool **67** and against the bias of control spring **76** moves the control spool **67** from right to left, closing with throttling edges **76a** the passage between the inlet chamber **64** and the supply chamber **24**.

With the pump **12** of a variable displacement type started up and with discharge line **62** blocked by valve spool **20**, in a well known manner, the pressure in the discharge line **62** will rise to a certain minimum pressure level, at which the differential pressure compensator **16** will move the displacement changing mechanism of the variable displacement pump **12** to a zero flow position and maintain the discharge line **62** at this minimum pressure level. Therefore only suction fluid, due to the internal leakage of the pump **12**, has to be delivered to inlet of the pump **12** and most of the flow will be delivered by the make-up pump **92** through line **91**, the heat exchanger **90**, lines **89** and **94**, annular space **102**, space **99** and line **107** to the reservoir **17**, at a minimum pressure level.

Assume that the load chamber **25** is subjected to a positive load. The initial displacement of the valve spool **20** to the right will connect the load chamber **25** through signal slot **41** with positive load sensing port **32**, white lands **21**, **22** and **23** still isolate the supply chamber **24**, load chambers **25** and **26** and outlet chambers **27** and **28**. As previously described positive load signal transmitted from positive load sensing port **32**, through signal passage **34**, check valve system and signal line **36** to the pump flow control **16** will increase the pressure in discharge line **62** to a level, which is higher by a constant pressure differential than the load pressure signal. The load pressure, transmitted through passage **72** to control space **71**, will move the positive load control spool **67** to the right, opening through slots **75** communication between the inlet chamber **64** and the supply chamber **24**. Communication will be maintained between the supply chamber **24** and the inlet chamber **64**, as long as the pump flow control **16** maintains a constant pressure differential between the pump discharge pressure and the positive load pressure. Positive load pressure signal from signal line **36** will also be transmitted through line **105** to space **101** where, reacting on the cross-sectional area of the piston **98**, will move it together with the spool **97**, against the biasing force of spring **106**, all the way to the right, cutting off communication between annular space **102** and space **99**. The fluid flow, delivered from the make-up pump **92**, will raise the pressure in the total exhaust system to the level, equivalent to pressure setting of the exhaust relief valve **60** and will pass through it to the system reservoir **17**. Therefore with the positive load sensing circuit of either of the flow control valves activated and the signal line **36** subjected to a pressure level, higher than that, equivalent to preload in the spring **106**, the exhaust unloading valve **95** will automatically isolate the fluid flow delivered by the make-up pump **92** from the reservoir **17**, thus diverting all of the flow from the exhaust circuit through exhaust relief valve **60**, automatically raising the pressure level of the exhaust circuit to that equivalent to the pressure setting of the exhaust relief valve **60**. The differential pressure compensator **16**, as is well known in the art, is always provided with leakage path to the reservoir **17**. For purposes of demonstration such a leakage path is shown diagramatically from signal line **36**. With positive load pressure signals from system valves no longer being transmitted, the pressure in the signal line **36** will drop to pressure of reservoir **17** and under action of the biasing spring **106** the spool **97** and the piston **98** will be moved all the way to the left, connecting exhaust circuit, through annular space **102**, with system reservoir. Under those conditions, with the exhaust circuit maintained at a minimum pressure level, exhaust flow from the make-up pump **101** will be bypassed to system reservoir **17** at a minimum pressure level.

Further displacement of the valve spool **20** to the right will connect the load chamber **25**, through metering slot **43**, with the supply chamber **24** and will also connect through metering slot **48** the load chamber **26** with the outlet chamber **28**. In a manner as previously described, the pump flow control **16** will maintain a constant pressure differential across orifice, created by displacement of metering slot **43**, the flow into the load chamber **25** and chamber **11b** of the fluid motor **11** being proportional to the area of the orifice and therefore displacement of the valve spool **20** from its neutral position and independent of the magnitude of the load L. Proportional fluid flow, larger by displacement of piston rod **11c**, will be delivered from the chamber **11a** and transferred through the load chamber **26**, metering slot **48**, outlet chamber **28**, slots **49** and the exhaust chamber **51** to the exhaust circuit. Since the outlet flow from the fluid motor **11** is larger than the inlet flow requirement of the pump **12**, the flow of fluid, passed through the relief valve **60** and equivalent to flow delivered by make-up pump **92**, will be increased by flow equivalent to displacement of the piston rod **11c**. With the pump flow at a controlled level delivered to the load chamber **26** and the chamber **11a**, proportional flow, smaller by displacement of the piston rod **11c**, will be delivered from the chamber **11b** to the exhaust system. Therefore the inlet flow into the fluid motor **11** is larger than its outlet flow, which must be supplemented by the flow from the make-up pump **92**, to maintain inlet of the pump **12**, to maintain inlet of the pump **12** pressurized. Therefore flow of fluid through the exhaust relief valve **60** will be reduced and equal to the difference between the flow delivered into the exhaust circuit by the make-up pump **92** and the flow, equivalent to displacement of the piston rod **11c**. During control of positive load the free floating piston **70** is subjected to pressure in the supply chamber **24** and through negative load sensing port **40** to the low pressure in the load chamber **26**. This pressure differential maintains the free floating piston **70** to the right closing with projection **78** and port **79** communication between control spaces **77** and **53**, effectively deactivating the negative load control spool **50**.

Assume that while controlling positive load L through the flow control valve **10**, a higher positive load W is actuated through the flow control valve **14**. Higher load pressure signal from the flow control valve **14** will be transmitted through the check valve system to the pump flow control **16**, which will now maintain system pressure, higher by a constant pressure differential, than pressure generated by positive load W. In a manner as previously described, the pressure drop through metering slot **43** will increase, therefore increasing the pressure differential between space **68** and control space **71**. The positive load control spool **67** will move into its modulating position, throttling with throttling edges **76***a* the fluid flowing from the inlet chamber **64** to the supply chamber **24**, to maintain a constant pressure differential between the supply chamber **24** and the load chamber **25**, thus controlling fluid flow through metering slot **43**. While this throttling control action takes place, control space **77** is connected through the negative load pressure sensing port **40** with low pressure existing in the load chamber **26**. The same low negative load pressure signal will be transmitted through line **103** to space **100**, where it will not affect the operation of the exhaust unloading valve **95**. Free floating piston **70**, subjected to pressure in the supply chamber **24** is maintained to the right and closes with projection **78** port **79**, leading to control space **53**. In this way negative load control spool **50** becomes isolated from the negative load pressure signal and the negative load control spool **50** must remain inactive during control of positive load. This action of the free floating piston **70** provides an effective interlock between positive and negative load controllers.

Assume that the load chamber **26** is subjected to a negative load L and that the valve spool **20** is displaced from its neutral position to the right while, as previously described, the positive load control spool **67** is maintained by the pump standby pressure in a position blocking communication between the inlet chamber **64** and the supply chamber **24**. Initial displacement of the valve spool **20** will connect through signal slot **41** the load chamber **25** with the positive load sensing port **32**. Since the load chamber **25** is subjected to low pressure neither the pump flow control **16** nor the positive load control spool **67** nor the exhaust unloading valve **95** will react to it. Simultaneously signal slot **47** will be connected to the negative load sensing port **40**, connecting the load chamber **26**, subjected to negative load pressure through signal passages with control space **77**. Since the control spool **67**, biased by control spring **76**, is contacting the free floating piston **70**, the pressure differential, developed between control space **71** and control space **77** will move the free floating piston **70** and the control spool **67** to the left, opening with projection **78** port **79**, cross-connecting control space **77** with control space **53**. Under action of negative load pressure, supplied from the negative load pressure sensing port **40**, the free floating piston **70** will move control spool **67** all the way to the left, isolating with throttling edges **76***a* the supply chamber **24** from the inlet chamber **64**. At the same time negative load pressure from control space **77**, transmitted through port **79** to control space **53**, reacting on the cross-sectional area of negative load control spool **50** will move it, against the biasing force of spring **54**, all the way to the right, with throttling edges **52** cutting off communication between the outlet chamber **28** and the exhaust chamber **51**. The negative load pressure from control space **77** will also be transmitted through line **103** to space **100** where, reacting on the cross-sectional area of spool **97**, will move it all the way to the right, against biasing force of spring **106**, cutting off direct communication between the discharge line of the make-up pump **92** and the reservoir **17**. Due to the pressure differential between spaces **100** and **101** the piston **98** will be maintained in the position as shown in the drawing. Therefore in the presence of either positive or negative load pressure signals, higher than that equivalent to preload in the spring **106**, the exhaust unloading valve **95** will automatically isolate the outlet of the make-up pump **92** from direct communication with the reservoir **17**, raising the pressure of the exhaust circuit to that, equivalent to the pressure setting of the exhaust relief valve **60**. Space **100** can be connected through a suitable leakage orifice in the spool **97** communicating with space **93**. Such an orifice, well known in the art, is diagrammatically shown on the drawing and would act in a similar way as clearance between the spool **97** and bore **96**. Then in the absence of the negative load pressure signal the spool **97**, biased by spring **106**, would move from right to left, to the position as shown in the drawing, providing a direct passage between the outlet of the make-up pump **92** and the reservoir.

Further displacement of valve spool **20** to the right will connect through metering slot **48** the load chamber **26** and the chamber **11***a* with the outlet chamber **28**, while also connecting through metering slots **43** the load chamber **25** with the supply chamber **24**. Since the outlet chamber **28** is isolated by position of the negative load control spool **50**, the pressure in the outlet chamber **28** will begin to rise, until it will reach a level, at which force generated on the cross-sectional area of the negative load control spool **50**, by the pressure in control space **53**, will equal the sum of the force generated on the same cross-sectional area by the pressure in the outlet chamber **28** and therefore pressure in space **56** and the biasing force of the spring **54**. At this point the negative load control spool **50** will move from right to left, into a modulating position, in which fluid flow from the outlet chamber **28** to the exhaust chamber **51** will be throttled by the throttling edges **52**, to automatically maintain a constant pressure differential, equivalent to biasing force of the spring **54**, between the load chamber **26** and the outlet chamber **28**. Since during control of negative load a constant pressure differential is maintained across the orifice, created by the displacement of metering slot **48**, by the throttling action of negative load control spool **50**, fluid flow through metering slot **48** will be proportional to the displacement of the valve spool **20** and constant for each specific position of metering slot **48**, irrespective of the change in the magnitude of the negative load L.

As previously described during control of negative load the control spool **67** will be maintained by the free floating piston **70** in a position, where it isolates the inlet chamber **64** from the supply chamber **24**. The inlet flow requirement of load chambers **25** and **26** and chambers **11***a* and **11***b* is supplied through check valves **83** and **84** from the outlet flow from one of the load chambers and total system exhaust flow available from the exhaust manifold, pressurized by the exhaust relief valve **60**. The pressure setting of the exhaust relief valve **60** is high enough to provide the necessary pressure drop through check valve **83**, at the highest rates of flow from the exhaust manifold to the load chamber **25**, without pressure in the load chamber **25** dropping below atmospheric level, thus preventing any possibility of cavitation. In this way, during control of negative load, inlet flow requirement of the actuator is not supplied from the pump circuit but from the pressurized exhaust circuit of flow control valves **10** and **14** and from the make-up pump **92**, conserving the pump flow and increasing system efficiency. If negative load pressure is not sufficiently high to provide constant pressure drop through metering slot **48**, the negative load control spool **50** will move to the left from its modulating and throttling position, the negative load pressure in the load chamber **26** and control space **77** will drop to a level at which the pressure in space **68**, due to the setting of the exhaust relief valve **60**, with the biasing force of control spring **76** will move the free floating piston **70** to the right together with the control spool **67** with projection **78** closing port **88**. The check valve **83** will close and the control system will revert to its positive load mode of operation, providing the energy to load L from the pump circuit to maintain a constant pressure differential across metering slot **43**, which will also maintain a constant pressure differential across metering slot **32**. During control of negative load the inlet flow requirement of the actuator is supplied from the outlet flow from the actuator, bypass flow from pump flow control, flow from the make-up pump **92** and the exhaust circuits of all of the other system flow control valves through check valves **83** and **84**. With discharge line **62** blocked and with the pump **12** being of fixed displacement type, the full pump flow is diverted at minimum pressure level by differential pressure bypass valve **16**, through line **80** into the exhaust circuit. When controlling a negative load from the chamber **11*a***, the inlet flow requirement of the chamber **11*b***, of fluid motor **11**, is less than outlet flow delivered from the chamber **11*a***. Therefore an excess flow, equivalent to displacement of the piston rod **11*c***, is passed into the exhaust circuit. Since the total inlet flow requirement of the pump **12** is supplied from its own bypass circuit, the total flow of the make-up pump **92** plus the flow, equivalent to the displacement of the piston rod **11*c***, will be passed through exhaust relief valve **60**. If the negative load is unidirectional and if it takes place in chamber **11*a*** signal line **103** can be blocked by plug **104**. Then, during control of negative load, the make-up pump **92** can be completely unloaded, by absence of the positive load signal and the inlet flow requirements of the pump **12** will be supplied directly from the chamber **11*a***, the exhaust relief valve **60** pressurizing the exhaust circuit and passing the volume of fluid within the displacement of the piston rod **11*c***. While the make-up pump **92** is unloaded the pressurized circuit is isolated from the outlet of the make-up pump **92** by the check valve **88**. In this way, with the make-up pump **92** unloaded, during control of negative load the heat input into the working fluid is reduced and the system efficiency increased.

When controlling a negative load from the chamber **11*b*** outlet flow of the fluid motor **11** is smaller than its inlet flow requirement into the chamber **11*a***. The difference between this inlet and outlet flow, equal to the flow, equivalent to the displacement of the piston rod **11*c***, is supplied into the chamber **11*a*** from the make-up pump **92**. Under those conditions the flow passing through the exhaust relief valve **60** is equal to the difference between output flow of the make-up pump **92** and the flow, equivalent to the displacement of piston rod **11*c***. The higher the speed of the load L the higher the flow required by displacement of the piston rod **11*c***. Therefore, when controlling a negative load from the chamber **11*b***, the maximum speed of the motor **11** is limited by the flow capability of the make-up pump **92**. When controlling a negative load from the chamber **11*a***, since the outlet flow exceeds the inlet flow requirement of the motor **11**, the exhaust circuit can be maintained pressurized at all speeds of load L, even without the presence of the make-up pump **92**.

During control of negative load, with valve spool **20** displaced to the left, the metering slot **46** throttles the oil flow to outlet chamber **27** and this flow is supplied through ports **29**, central passage **30** in valve spool **20** and ports **31** to the outlet chamber **28**. Therefore ports **29**, central passage **20** and ports **31** cross-connect outlet chambers **27** and **28** permitting bidirectional control of negative load.

The embodiment of flow control valve **10** is such that it provides a load responsive valve with positive load metering orifices between the supply chamber and the load chambers and negative load metering orifices between the load chambers and the outlet chambers. Therefore the positive load control responds to socalled upstream pressure differential, well known in the art. There are other types of load responsive valves, which in control of positive load, respond to down stream pressure differential or down stream pressure. Such load responsive controls are disclosed in applicant's U.S. Pat. Nos. 3,984,979 of Oct. 12, 1976, 3,998,134 of Dec. 21, 1976 and 4,099,379 of July 11, 1978. In those load responsive controls no throttling action is present between the supply chamber and the load chambers and therefore the positive load sensing ports are directly subjected to the pressure of the supply chamber, with the valve spool moved from its neutral position. Since load chambers of all of those valves are always maintained at a certain minimum pressure level, this pressure can be used to actuate the exhaust unloading valve **95**, during control of negative load. Therefore with those load responsive valves the negative load pressure input through line **103** to space **100** is not necessary and the exhaust unloading valve **95** will be activated just by the signal from the positive load sensing ports, during control of positive and negative loads, once the valve spool is moved in either direction from its neutral position.

During control of negative load, with the pump **12** being of a variable displacement type, due to the blocked outlet the pump is maintained in zero displacement position, with zero inlet flow requirement. When controlling a negative load from the chamber **11*a***, in a manner as previously described, excess flow is generated into the exhaust circuit. If the negative load is unidirectional and if it is generated in the chamber **11*a***, the plug **104** can be used and as previously stated the make-up pump **92** can remain unloaded during control of negative load, the check valve **88** isolating the pressurized exhaust circuit from the make-up pump **92**. When controlling the negative load from chamber **11*b*** the flow, equivalent to the displacement of the piston rod **11*c***, must be supplied into the fluid motor **11** from the exhaust circuit, this flow being supplied by the make-up pump **92**. Since the inlet flow requirement, due to displacement of the piston rod **11*c***, is proportional to speed of the load L, it is therefore limited by the maximum flow capacity of the make-up pump **92**.

With flow control valves **10** and **15** controlling loads L and W, during control of negative loads, in a manner as previously described, the pump flow is not used to supply the inlet flow requirement of the fluid motors, but is completely blocked from the fluid motors, the motor outlet flow supplying its own inlet flow requirement. Under those conditions the make-up pump **92** is not necessary to supply the inlet pressurization of the system pump, but to supply directly, if required, the make-up flow into the system motors. In the load responsive system of the present invention the make-up pump **92** performs the dual function of providing the necessary pressurization of the system pump inlet and of providing the necessary pressure for flow exchange between ports of the fluid motors, during control of negative loads. Since the system pump supplies the energy into the system, proportional to pressure differential existing across its ports, the energy of the pressurized exhaust fluid, from the system motors, is recovered thus improving the system efficiency.

The flow control valve **10**, with its free floating piston **70** blocked, will perform in a different way. During control of negative load from the chamber **11*b***, or when controlling a positive load using the chamber **11*a*** the flow, equivalent to the displacement of the piston rod **11*c*** must be supplied into inlet port of pump **12** from the make-up pump **92**. When controlling a positive load and connecting the pump **12** to the chamber **11*b***, the outlet flow from the chamber **11*a*** exceeds the inlet flow requirement of the pump by flow, equivalent to the displacement of the piston rod **11*c***. Under these conditions the make-up pump **92** could be unloaded and the system efficiency increased. This can be done by providing an additional positive load sensing port **33** and connecting it either directly, or through a separate check valve logic system to space **101**, while the line **105** is blocked. Under those conditions, during control of positive load, the make-up pump **92** would be unloaded when connecting the system pump with the chamber **11*b*** and would be automatically activated when supplying the fluid flow into the chamber **11*a***. In this way the system efficiency can be further improved. The same approach can be taken, while controlling a negative load. By providing an additional negative load sensing port **39** and connecting it directly to space **100**, only during the presence of negative load in the chamber **11*b*** would the exhaust unloading valve **95** be activated and the make-up pump **92** would supply fluid flow into the exhaust circuit. When controlling negative load from the chamber **11*a*** the make-up pump **92** would be completely unloaded, increasing system efficiency.

The exhaust unloading valve **95** is made responsive to both positive and negative load pressure signals and permits pressurization of the exhaust circuit, when either positive or negative loads are being controlled, providing the pressurized inlet flow either to the system pump or directly to the system motors. With the positive and negative load pressure signal dropping below a certain minimum predetermined level, signifying that the system valves are in neutral and the system pump is in a standby condition, the outlet of the make-up pump **92** is automatically connected by the exhaust unloading valve **95** directly to system reservoir **17**. Under those conditions the make-up pump **92** is completely unloaded, greatly reducing the system loss in standby condition. The exhaust system is automatically unloaded only under the conditions, where, for example, a variable displacement pump is in its zero displacement position, not requiring pressurization of its inlet, or during control of unidirectional negative load from chamber **11*a*** and negative load pressure signal blocked by the plug **104**.

The exhaust unloading valve **95**, with its spool **97** and piston **98**, is directly operated by the energy derived from the negative and positive load sensing circuits. To minimize the use of this energy, in a well known manner, the spool **97** and piston **98** may be operated by a pilot valve, responsive to the positive and negative load pressure signals, but using energy from an external pressure source. In this way a minimal amount of energy from the load sensing circuits can be used in actuation of the flow controlling elements of the exhaust unloading valve **95**.

The exhaust unloading valve **95**, actuated by the load pressure signals, can also be used to perform another function. With system pump **12** being of a fixed displacement type and pump control **16** being a differential pressure relief valve, the annular space **102** can be directly connected to discharge line **62**. Under those conditions the exhaust unloading valve **95** will act as an unloading valve of the differential bypass valve **16**, ensuring that with valve spool in neutral position and load pressure signals at minimum level, the total pressure circuit of the pump is fully unloaded, with discharge line of the pump being directly connected by exhaust unloading valve **95** to the system reservoir **17**.

Although the preferred embodiment of this invention has been shown and described in detail it is recognized that the invention is not limited to the precise form and structure shown and various modifications and rearrangements as will occur to those skilled in the art upon full comprehension of this invention may be resorted to without departing from the scope of the invention as defined in the claims.

What is claimed is:

1. A valve assembly supplied with pressure fluid by a pump having inlet port means and outlet port means, said valve assembly comprising a housing having a fluid inlet chamber connected to said outlet port means, at least one load chamber, and exhaust means connected to reservoir means, first valve means for selectively interconnecting said load chamber with said inlet chamber and said exhaust means, signal port means in said housing selectively communicable with a source of pressure by said first valve means, charging pump means having discharge port means connected to said inlet port means of said pump and suction port means connected to said reservoir means, check valve means interposed between said inlet port means and said discharge port means and operable to connect for one way fluid flow said charging pump means and said inlet port means and second valve means interposed between said discharge port means and said reservoir means upstream of said check valve means having means responsive to pressure in said signal port means and means to unload said charging pump means.

2. A valve assembly as set forth in claim **1** wherein said exhaust means has fluid connecting means to said inlet port means of said pump.

3. A valve assembly as set forth in claim **2** wherein said exhaust means has exhaust relief valve means.

4. A valve assembly as set forth in claim **1** wherein said inlet port means of said pump has exhaust relief valve means connected to said reservoir means.

5. A valve assembly as set forth in claim 1 wherein said first valve means has means operable to selectively communicate said signal port means and said load chamber.

6. A valve assembly as set forth in claim 1 wherein said first valve means has means operable to selectively communicate said signal port means and said inlet chamber.

7. A valve assembly as set forth in claim 1 wherein said means responsive to pressure in said signal port means has disconnecting means operable to disconnect said charging pump means of said pump from said reservoir means when pressure in said signal port means exceeds a certain predetermined level.

8. A valve assembly as set forth in claim 1 wherein said means responsive to pressure in said signal port means has connecting means operable to connect said charging pump means to said reservoir means when pressure in said signal port means drops below a certain predetermined level.

9. A valve assembly as set forth in claim 1 wherein said signal port means includes positive load pressure port means communicable with an output flow control of said pump and with said means to unload said charging pump means.

10. A valve assembly as set forth in claim 1 wherein said signal port means includes negative load pressure port means communicable with said means to unload said charging pump means.

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